SUBCOOLING LIQUID REFRIGERANT 12 IN A VAPOR COMPRESSION REFRIGERATION SYSTEM WITH A LIQUID-VAPOR HEAT EXCHANGER

by

JOHN D. CHRISTIE

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Signature of Author . . . . John D. Christie . . . .
Department of Mechanical Engineering, May 25, 1959

Certified by . . . . A. L. Hesselschwerdt, Jr.
Thesis Supervisor

Accepted by . . . . B. G. Rightmire . . . .
Chairman, Departmental Committee on Theses

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Department of Mechanical Engineering
Massachusetts Institute of Technology
Cambridge 39, Massachusetts
May 25, 1959

Professor Alvin Sloane
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

Submitted herewith is a thesis entitled "Subcooling Liquid Refrigerant 12 in a Vapor Compression Refrigeration System with a Liquid-vapor Heat Exchanger" in partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering.

Respectfully submitted,

John D. Christie

John D. Christie

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I would like to express my appreciation and gratitude to Professor A. L. Hesselschwerdt, Jr. for his helpful guidance and to Mr. B. Dimond for his assistance in the laboratory while I was preparing this thesis.

Sincerely,

John D. Christie

John D. Christie

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ABSTRACT

Subcooling liquid refrigerant in a vapor compression refrigeration cycle increases the system capacity. The purpose of this thesis is to relate theoretical and experimental performance characteristics to one method of subcooling Refrigerant 12, the use of a liquid-vapor heat exchanger.

Theoretical and experimental values of system coefficient of performance and horsepower per ton are found at various evaporating pressures for a system with and without a liquid-vapor heat exchanger. The theoretical performance of a system with an evaporating pressure of 26 psia and a condensing pressure of 110 psia is found for variations in heat exchanger effectiveness between 0 and 1.

Theoretically a liquid-vapor heat exchanger has a negligible effect on system coefficient of performance and horsepower per ton. Experiments on a system with a relatively short suction line showed that the use of a liquid-vapor heat exchanger does not increase the system performance enough to warrant its installation in a system for this reason. The system capacity is increased by installing a liquid-vapor heat exchanger.

Any future work done with liquid-vapor heat exchangers for the purpose of increasing performance in Refrigerant 12 systems should be limited to systems with long suction lines.

I. INTRODUCTION

Subcooling liquid refrigerant before the expansion valve in a vapor compression refrigeration cycle increases the evaporator capacity for a given evaporating pressure. If the vapor refrigerant leaving the evaporator is heated before it enters the compressor, the volume of the vapor is increased and more work is required to compress the vapor to a given pressure. When liquid refrigerant is subcooled in a liquid-vapor heat exchanger (heat exchanger between the liquid and vapor refrigerant), both of these effects occur. One purpose of this thesis is to theoretically predict the changes in system performance that occur when a liquid-vapor heat exchanger is used to subcool liquid Refrigerant 12. The theoretical performance of the system with a liquid-vapor heat exchanger is compared with the performance of the system without a heat exchanger. This comparison is made over a range of operating conditions.

In an actual system under standard operating conditions energy losses occur in the motor, compressor, evaporator, and piping. These losses decrease the system performance below the theoretical predictions. Under average conditions with the piping exposed to room air the vapor refrigerant in the suction line will gain heat from the atmosphere. This heat flow increases the specific volume of the refrigerant before compression and decreases the system performance. If an equivalent heat flow to the vapor refrigerant in the suction line can be taken from the liquid refrigerant leaving the condenser rather than from the ambient air, the

liquid refrigerant can be subcooled without any added loss in performance. The increase in performance due to subcooling the liquid in a liquid-vapor heat exchanger is dependent upon the magnitude of the heat flow to the suction line when a heat exchanger is not used. If the ambient air temperature is high or the heat transfer surface of the suction line is large, the increase in performance that can be obtained with a heat exchanger may warrant its installation.

Another purpose of this thesis is to compare the actual performance of a Refrigerant 12 system with a liquid-vapor heat exchanger and with no heat exchanger. This comparison is made over a range of operating conditions using Refrigerant 12 in the system. These experimental results are also compared with the theoretical calculations for the same conditions.

II. THEORETICAL ANALYSIS

Theoretical calculations were made for a Refrigerant 12 compression cycle with a liquid-vapor heat exchanger. A counterflow heat exchanger was assumed for these calculations because the maximum amount of heat transfer is possible with this type of heat exchanger. The average specific heat of Refrigerant 12 vapor is less than that of the Refrigerant 12 liquid. Therefore the heat exchanger effectiveness for this case is defined as:

$$\epsilon = \frac{t_{vo} - t_{vi}}{t_{lo} - t_{li}}$$

When the effectiveness is 1, the temperature of the vapor out of the heat exchanger (t_{vo}) equals the temperature of the liquid entering the heat exchanger (t_{li}) and the maximum heat transfer occurs.

The coefficient of performance (COP) and horsepower per ton ($\frac{HP}{ton}$) were calculated for the cycle while the heat exchanger effectiveness (ϵ) was varied between 0 and 1. Figure 1 is a plot of coefficient of performance versus temperature of liquid leaving the heat exchanger (t_{lo}). The plot shows that the maximum possible increase in COP is about 2 per cent. Figure 2 is a plot of $\frac{HP}{ton}$ versus t_{lo}. The maximum decrease in $\frac{HP}{ton}$ is also about 2 per cent.

For all practical purposes the theoretical coefficient of performance and horsepower per ton for a cycle operating between these pressures is approximately constant. This is also shown to be true for other evaporating pressures. Further theoretical calculations were made using experimentally determined values for evaporating pressure, condensing pressure, and heat exchanger effectiveness. These results are shown in Figure 14, a plot of coefficient of performance versus evaporating pressure for cycles with and without a liquid-vapor heat exchanger. All the points in this

plot do not lie on two smooth curves because the condensing pressure varies between 109 psia and 121 psia. The important thing is that the theoretical COP is approximately the same for a cycle with a heat exchanger as it is for a cycle operating between the same pressures without a heat exchanger.

Theoretical calculations are shown in Appendix A with the corresponding assumptions for these calculations.

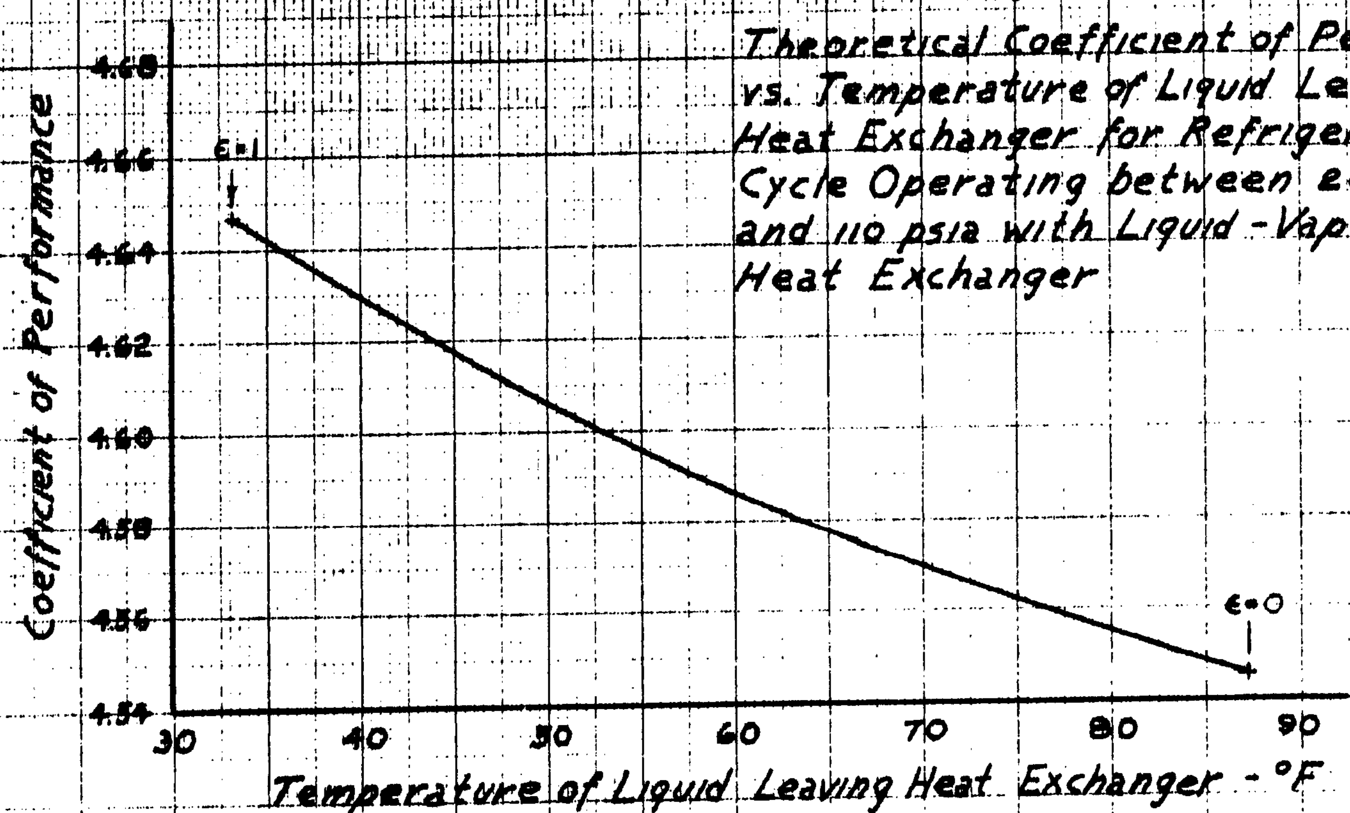

Figure 1

Theoretical Coefficient of Performance vs. Temperature of Liquid Leaving Heat Exchanger for Refrigerant 12 Cycle Operating between 26 psia and 110 psia with Liquid-Vapor Heat Exchanger

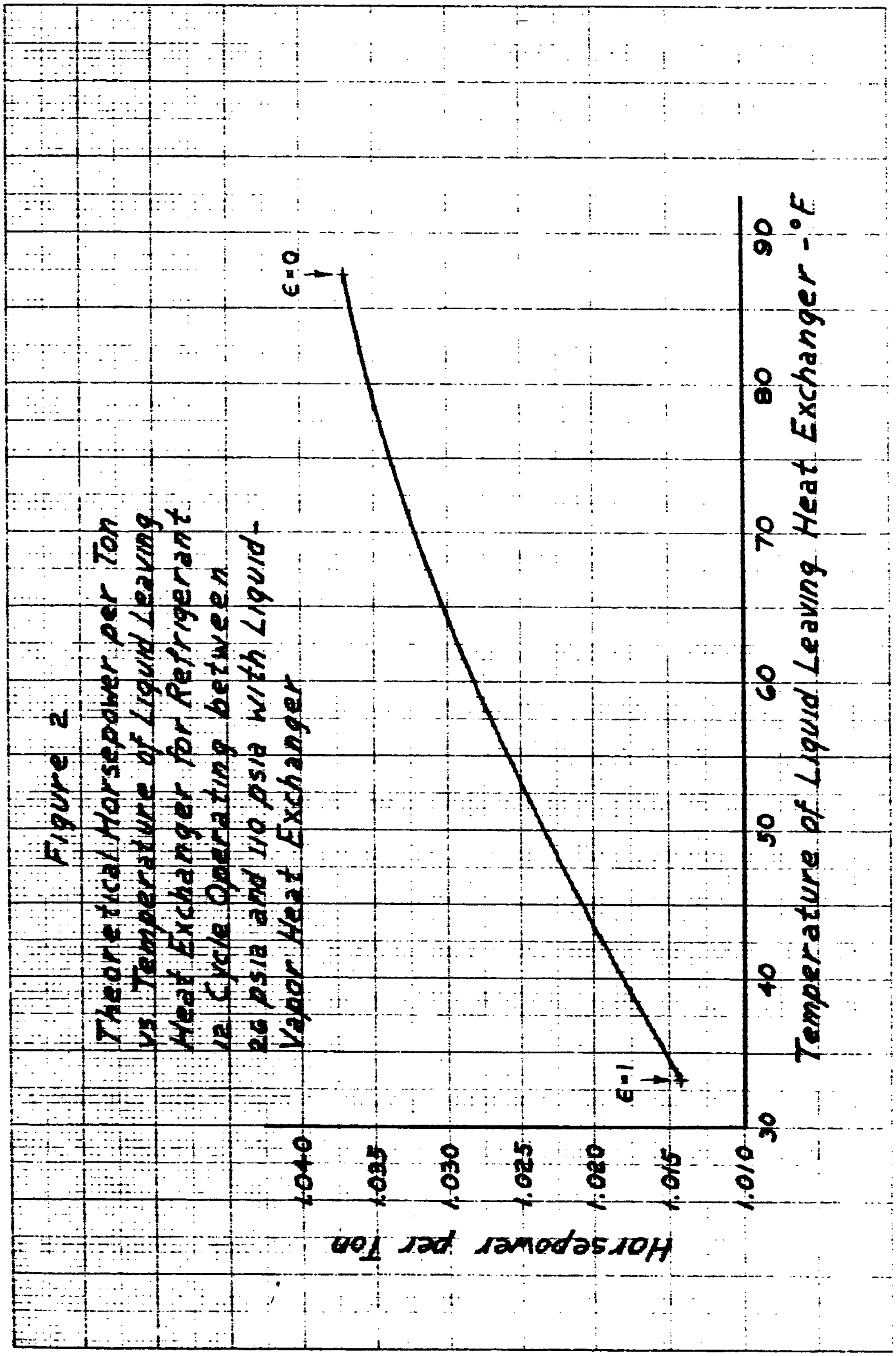

Figure 2

Theoretical Horsepower per Ton vs Temperature of Liquid Leaving Heat Exchanger for Refrigerant 12 Cycle Operating between 26 psia and 110 psia with Liquid-Vapor Heat Exchanger

III. EXPERIMENTAL APPROACH AND PROCEDURE

The purpose of making experimental runs was to obtain actual system characteristics to compare with theoretical results and to compare actual system characteristics with and without a liquid-vapor heat exchanger.

Two different sets of runs were made. The first 9 runs were made with the liquid-vapor heat exchanger in the system. Then 5 runs were made without the heat exchanger. The first set of runs was made with the heat exchanger so the pressure drop across the heat exchanger in the suction line could be measured. When the heat exchanger was removed, an equivalent length of piping was installed to provide an equal pressure drop between the same two points in the suction line. The system characteristics are then compared for approximately the same pressure drops in the suction line with and without the heat exchanger.

In all experimental runs a York Condensing Unit and a primary refrigerant calorimeter were used. The heat exchanger used in the first 9 runs was a single pass counter flow exchanger. Mineral wool blanket insulation was used to insulate the heat exchanger and all piping except the liquid line to the heat exchanger for the first 9 runs. This insulation was approximately one inch thick and was covered with aluminum foil to minimize heat transfer to and from the surroundings. The use of insulation improves the effectiveness of the heat exchanger and the accuracy of the heat exchanger effectiveness calculations.

A complete description of all equipment used is listed in Appendix B. A schematic diagram of the apparatus with the liquid-vapor heat exchanger is shown in Figure 3. The locations of all the data points are shown with their corresponding symbols. A photograph of the apparatus as it was set up for the first 9 runs is shown in Figure 4. Figure 5 is a photograph of the insulated heat exchanger showing the location of the pressure taps

and the tubing leading to a mercury manometer.

The heat exchanger and all the insulation were removed for the last 5 runs. An equivalent length of 12.5 feet of 5/8 inch O.D. refrigeration tubing was installed between the two pressure taps. The piping was not insulated for these runs in an attempt to approach normal operating conditions. Other than the extra length of suction line and the lack of insulation, the apparatus was the same for the second set of runs.

All runs were made after the equipment had been operating for one or more hours to reach steady state conditions. When the equipment reached steady state, data points were taken every five minutes for 45 minutes. The ten readings were then averaged before corrections and calculations were made. By taking an average of ten readings, errors due to small fluctuations in data and the taking of data were reduced.

All temperatures were measured with copper consantan thermocouples inserted in glands which were located in the fluid flow. Temperature corrections were made using a standard calibration curve. All pressures except the barometric pressure and the pressure drop across the heat exchanger were measured with bourdon gages. All pressure gages were calibrated before any experimental runs were made and corrections were made during these calibrations. The pressure drop across the heat exchanger or equivalent length of piping was measured with a mercury manometer. The electrical input to the calorimeter and motor were measured with wattmeters and corrected with calibration curves.

The tabulated average data for all runs is shown in Table I, Appendix C.

To compare the characteristics under different operating conditions, the actual COP, $\frac{hp}{ton}$, and evaporator capacity were calculated for each run. The heat exchanger effectiveness is calculated for each run were it was used. In calculating the COP, $\frac{hp}{ton}$, and evaporator capacity, the heat

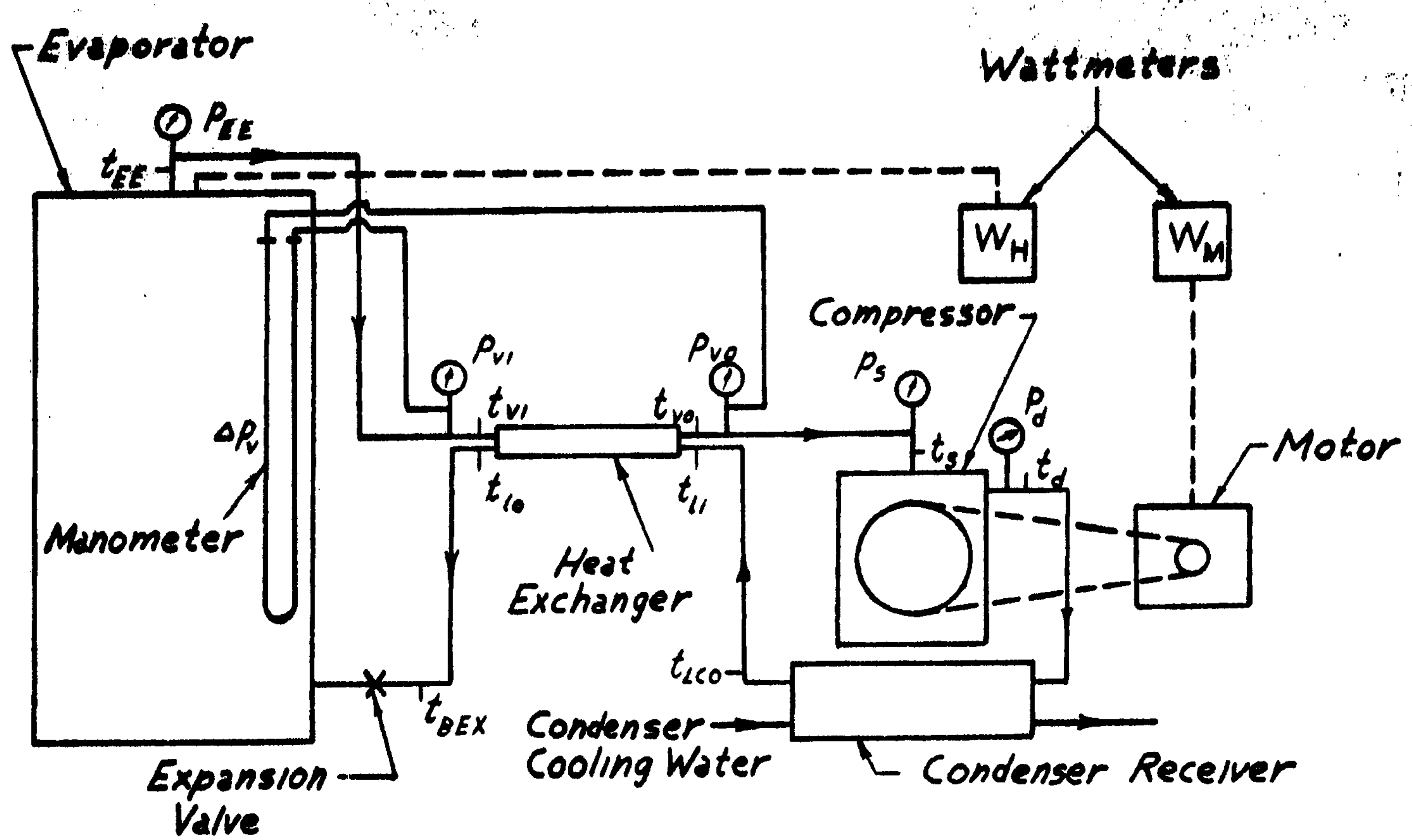

Figure 3
Schematic Diagram of Apparatus with Liquid - Vapor Heat Exchanger

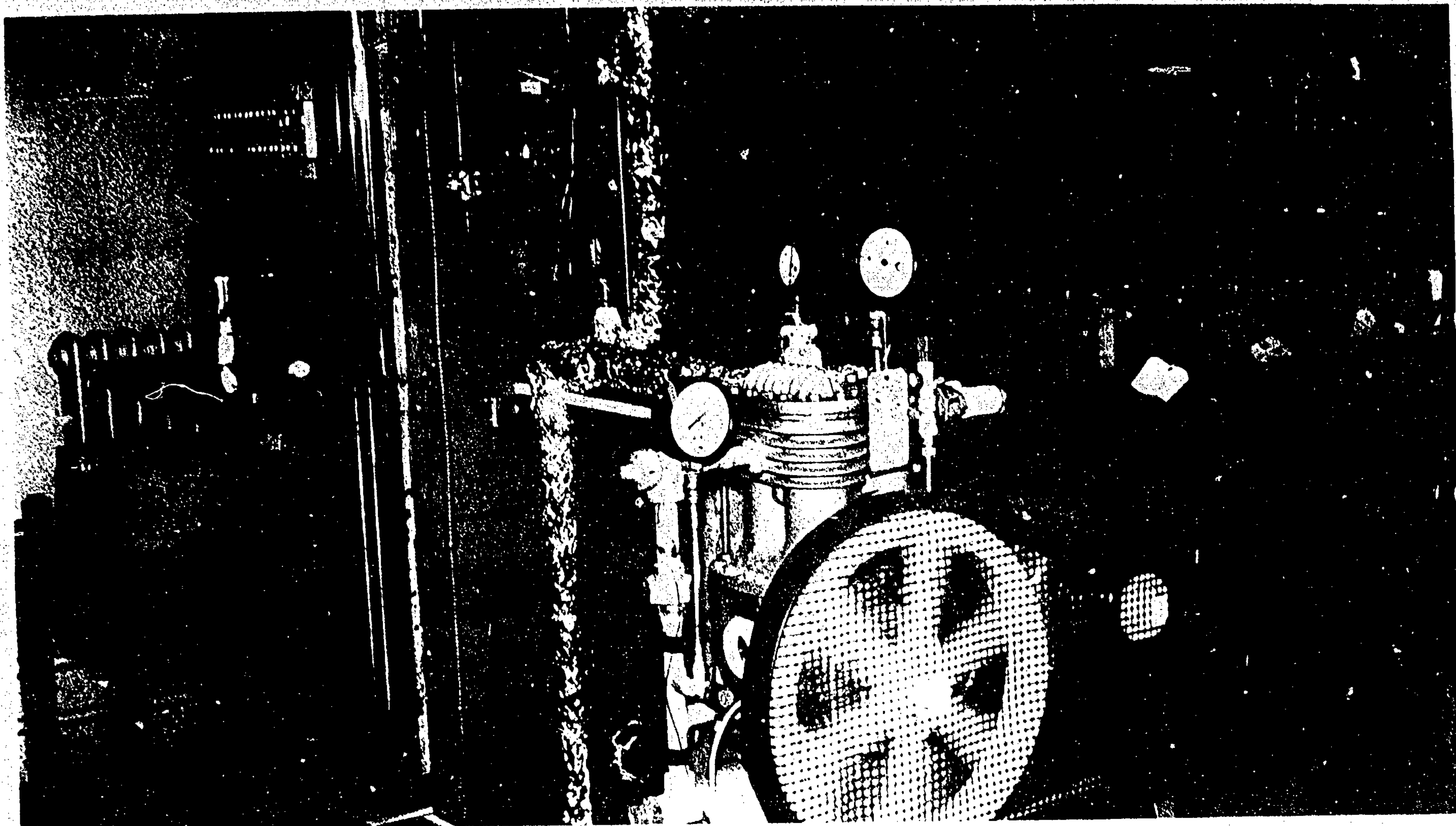

Figure 4. Experimental apparatus including liquid-vapor heat exchanger.

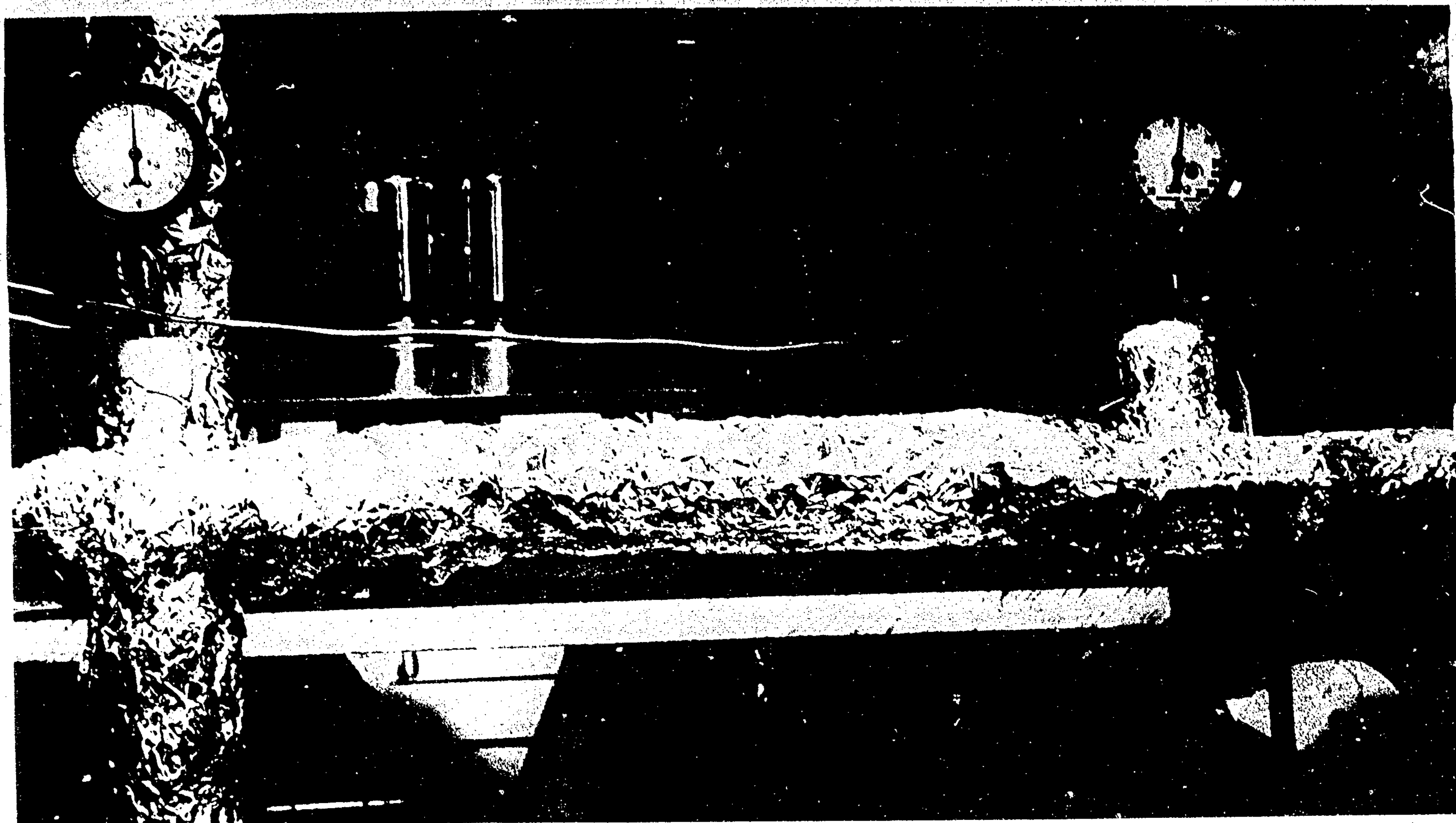

Figure 6. Insulated liquid-vapor heat exchanger as installed in apparatus.

transfer to the evaporator from the surroundings is assumed to be negligible compared to the electrical input. This is a good assumption, especially when the evaporating temperature is relatively high.

After calculations were made for the actual performance, the experimental heat exchanger effectiveness was used to make theoretical calculations of COP and $\frac{HP}{ton}$ for a system operating between the same evaporating and condensing pressures.

For the second set of runs without the heat exchanger, the equivalent length of piping to be installed was calculated. From the equation for pressure drop in a pipe:

$$p = \frac{0.0121\ fLw^2v}{d^5}$$

the equivalent length of piping (L) was determined. The average L for a number of calculations was 12.5 feet.

Similar experimental and theoretical calculations were made for the second set of runs without the heat exchanger. The theoretical calculations with no heat exchanger are the same as those with a heat exchanger where $\epsilon = 0$.

Details of the calculations are shown in Appendix D.

IV. RESULTS AND DISCUSSION OF RESULTS

Figure 1 shows the theoretical increase in COP that is possible by using a heat exchanger in a Refrigerant 12 cycle operating between 26 psia and 110 psia. Figure 2 is a similar plot showing the effect of the liquid-vapor heat exchanger on $\frac{HP}{ton}$. In both cases the maximum possible increase in performance is about 2 per cent. Similar results are shown for other evaporating pressures in Figure 14. If operating conditions were approximately ideal and the COP and $\frac{HP}{ton}$ were the most important characteristics of the system, the use of a liquid-vapor heat exchanger in the system would not be warranted. The increase in performance gained by subcooling the liquid Refrigerant 12 is not large enough to overcome the decrease in performance due to heating the vapor before the compressor and still cause an appreciable change on system COP.

Although the system COP and $\frac{HP}{ton}$ do not change appreciably when a heat exchanger is used, the evaporator capacity will increase. If a given evaporator is in a system and the tonnage of the system has to be increased, it might be possible to do so by adding a liquid-vapor rather than installing a larger evaporator.

The experimental and theoretical COP for a system with a liquid-vapor heat exchanger are plotted versus evaporating pressure (p_E) in Figure 6. All the theoretical points do not lie on a smooth curve because the condensing pressure (p_c) varies between 110 psia and 121 psia. The experimental curve follows the same trend as the theoretical curve but the deviation between the two is greater at higher values of p_E. This is reasonable because the refrigerant flow is greater at higher values of p_E. Consequently the pressure drops in the system are larger, causing a greater decrease in COP.

Figure 7 is a plot of theoretical and experimental $\frac{HP}{ton}$ versus p_E for

the system with the liquid-vapor heat exchanger. Both curves have the same trend with the experimental $\frac{HP}{ton}$ being at least twice as great as the corresponding theoretical value. The deviation between the two is greater at lower values of p_E. For lower values of p_E the specific volume of the refrigerant is greater, and changes in specific volume are greater for a given p_E. The specific volume of the refrigerant affects the actual work of compression and the efficiency of the compressor. The effect of specific volume is greater at lower values of p_E; therefore, the effect on the $\frac{HP}{ton}$ required is greater.

Heat exchanger effectiveness is plotted versus evaporating pressure in Figure 8. Theoretically ϵ is a function of heat exchanger geometry only. The experimental plot of ϵ versus p_E shows that ϵ decreases as p_E increases. As p_E increases the mean temperature difference between the two fluids in the heat exchanger decreases. The mass flow rate of the refrigerant through the heat exchanger increases as p_E increases. The actual heat exchanger effectiveness ϵ is dependent on the mean temperature difference and/or the mass flow rate of the fluids. The heat loss from the heat exchanger to the ambient air decreases as the heat exchanger effectiveness increases. If no heat losses occurred, the variation of ϵ would be smaller for the same variation of p_E.

Two points on the experimental plot deviate considerably from the curve in the figure. The high value of $\epsilon = 0.519$ was obtained when the heat loss from the heat exchanger was unusually low. The value of $\epsilon = 0.179$, when $p_E = 59.22$ psia, was obtained during a run when the system was not quite at a steady state condition. The temperature of the vapor in the heat exchanger varied over 4° F during the run. These two experimental values are known to be in error; therefore, they were not used to draw the curve.

Figure 9 is a plot of theoretical and experimental COP for the system

with the extra suction line and no heat exchanger. The curves follow the same trends as the ones for the system with the liquid-vapor heat exchanger. The deviation is again greater at higher values of p_E because of increased pressure drops.

The $\frac{HP}{ton}$ of the system without the heat exchanger is plotted versus p_E in Figure 10. The trends are the same for the system without the heat exchanger as they are for the system with the heat exchanger.

The actual pressure drop across the heat exchanger and the extra suction line for the two different sets of runs is plotted versus evaporating pressure in Figure 11. The Δp_v across the extra suction line is larger than the Δp_v across the heat exchanger for the same evaporating pressure. This deviation shows that the wrong length of tubing was installed for the extra suction line. This error in installing the wrong length of pipe is due to unaccounted pressure losses in the fittings. An equivalent length of pipe was determined for each fitting but the results were not accurate. The maximum deviation between the two curves is approximately 0.2 psi. This deviation is not large enough to cause any considerable error in comparing the experimental results for the two sets of runs.

The actual COP for the two different sets of runs is plotted versus p_E in Figure 12. For any p_E the system COP is higher when the liquid-vapor heat exchanger is in the system. The absolute increase in performance is about the same for any p_E, but the percentage increase in COP decreases as p_E increases.

A similar plot comparing $\frac{HP}{ton}$ for the two sets of runs is shown in Figure 13. The $\frac{HP}{ton}$ for the system with the heat exchanger is lower for any p_E. The deviation between the curves increases at lower values of p_E because of the increasing effect of specific volume changes.

Figure 14 is a plot of theoretical COP versus p_E for the system runs

with and without the heat exchanger. The points do not lie on smooth curves because of variations in condensing pressure. The plot shows that the difference in theoretical performance for systems with and without the liquid-vapor heat exchanger is negligible.

All tabulated results are shown in Table 2, Appendix E.

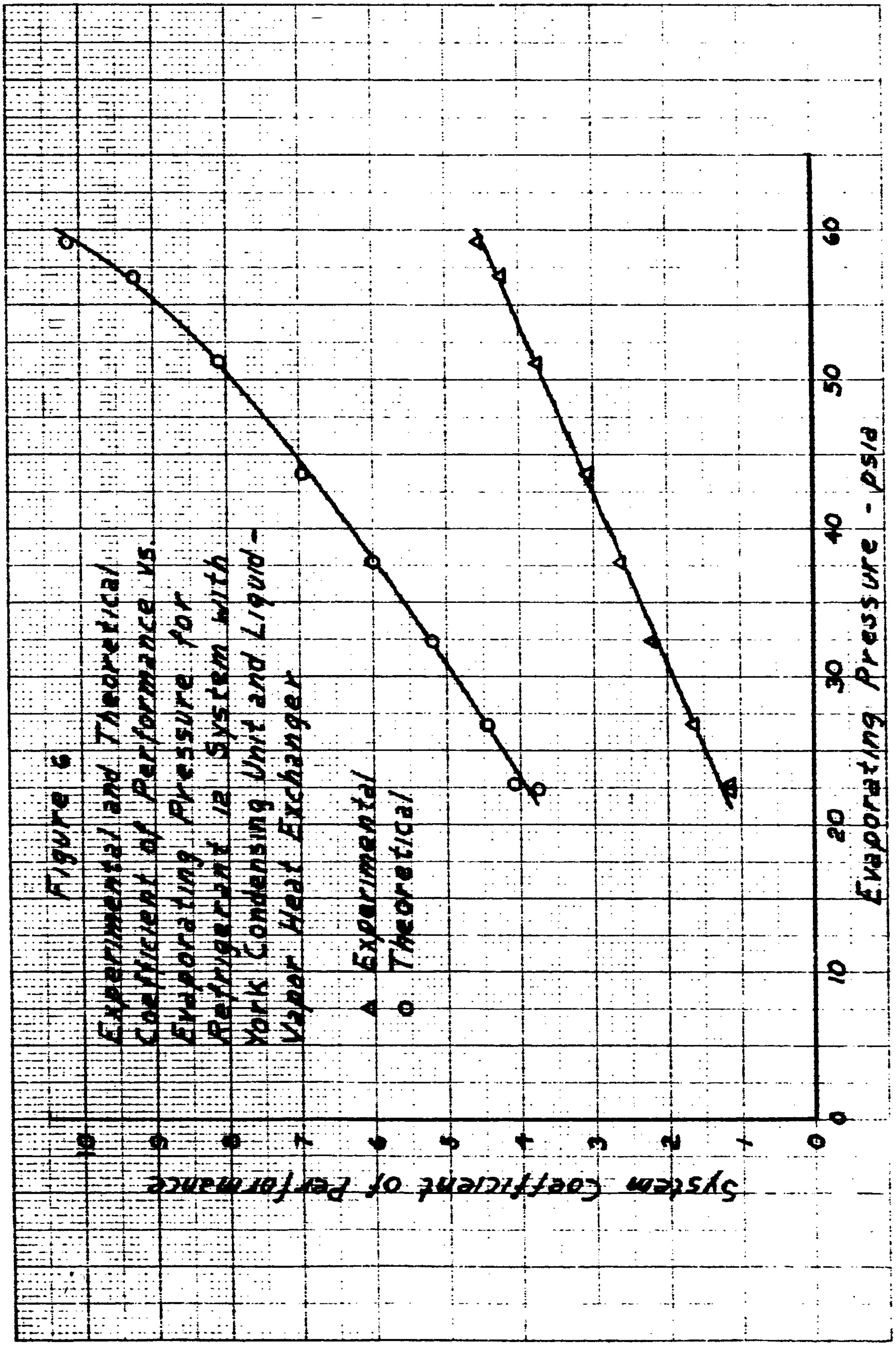

Figure 6
Experimental and Theoretical Coefficient of Performance vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit and Liquid-Vapor Heat Exchanger

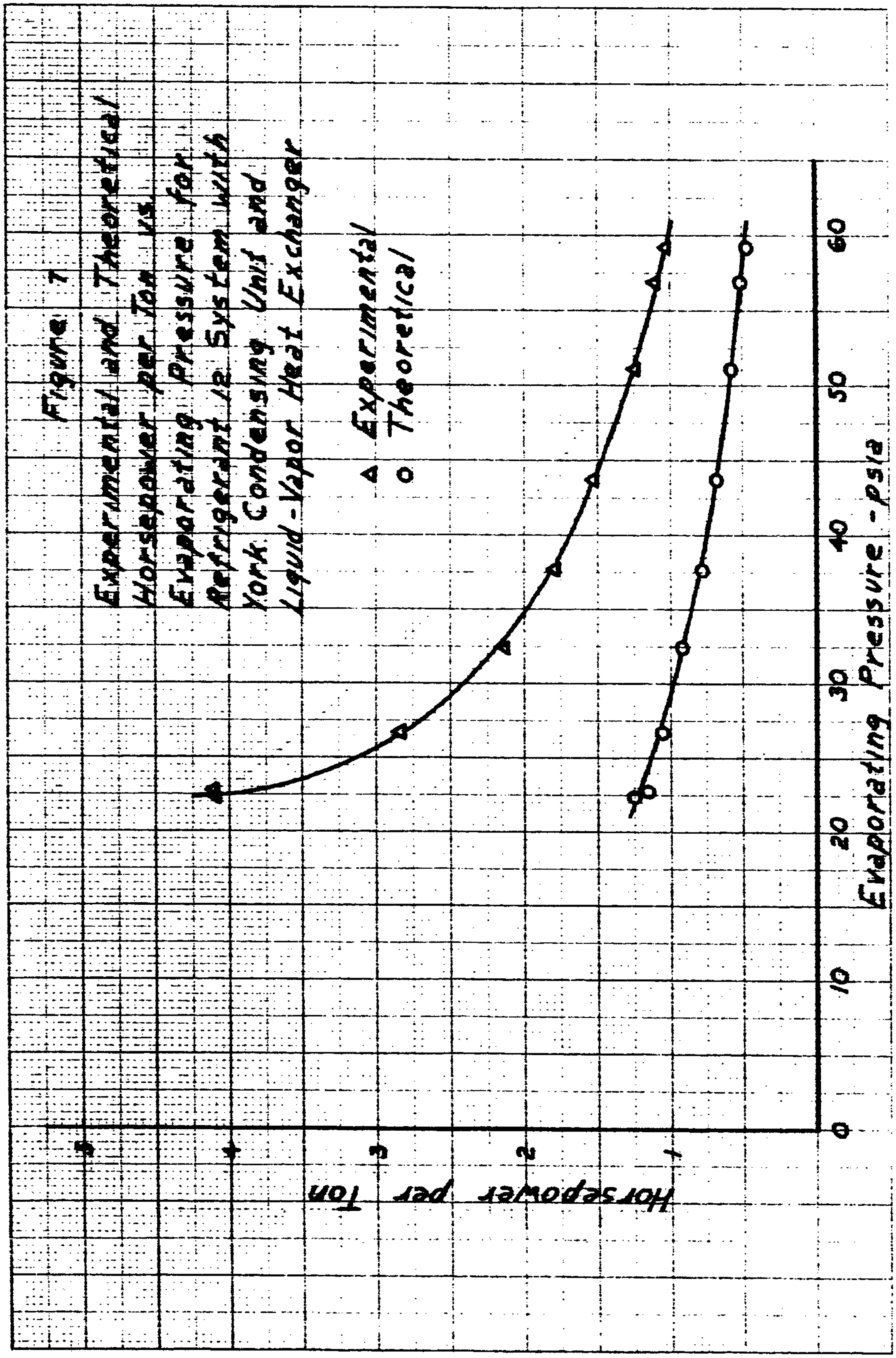

Figure 7
Experimental and Theoretical Horsepower per Ton vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit and Liquid-Vapor Heat Exchanger

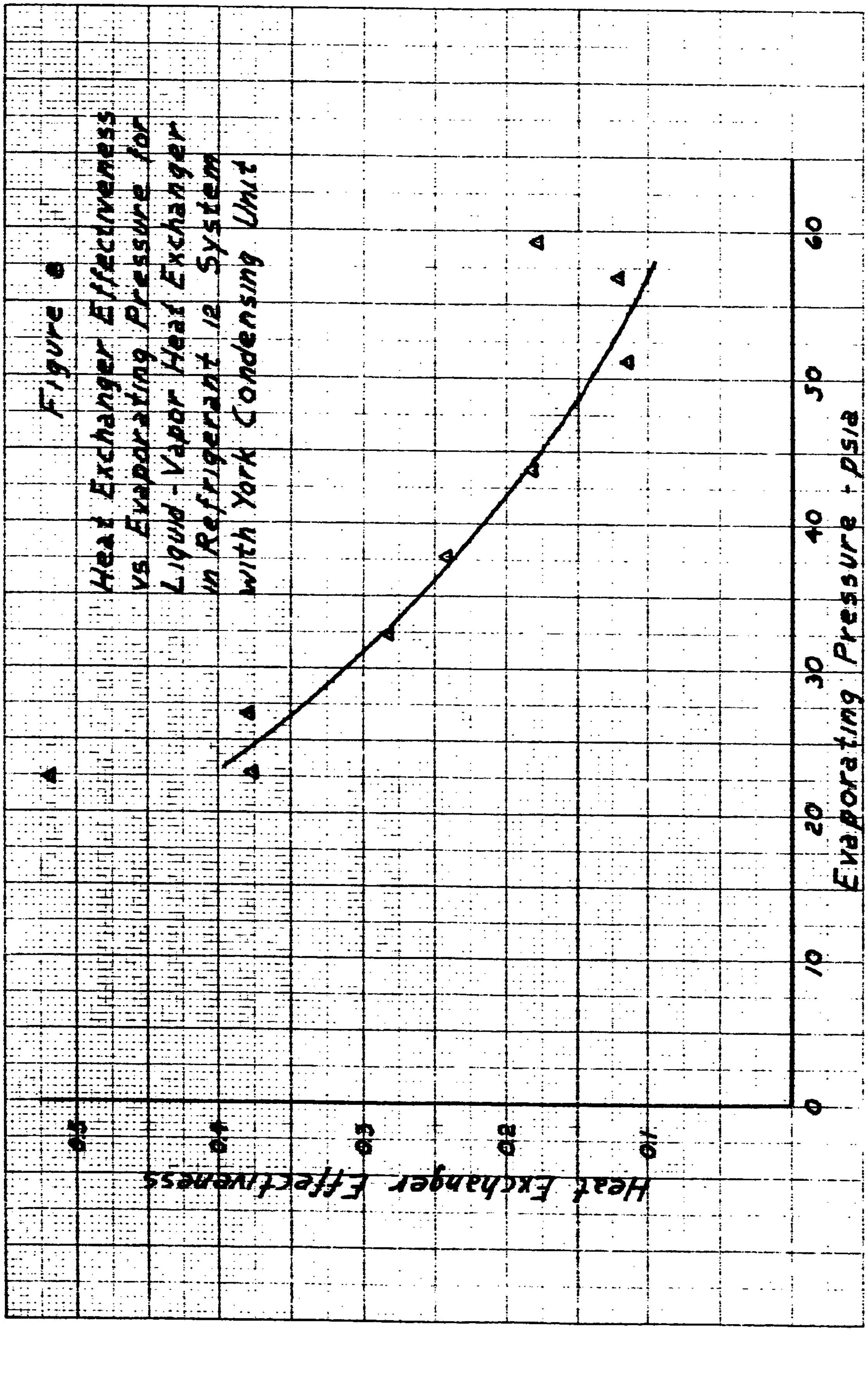

Figure 8
Heat Exchanger Effectiveness vs. Evaporating Pressure for Liquid-Vapor Heat Exchanger in Refrigerant 12 System with York Condensing Unit
Heat Exchanger Effectiveness
0.5
0.4
0.3
0.2
0.1
Evaporating Pressure - psia
0
10
20
30
40
50
60

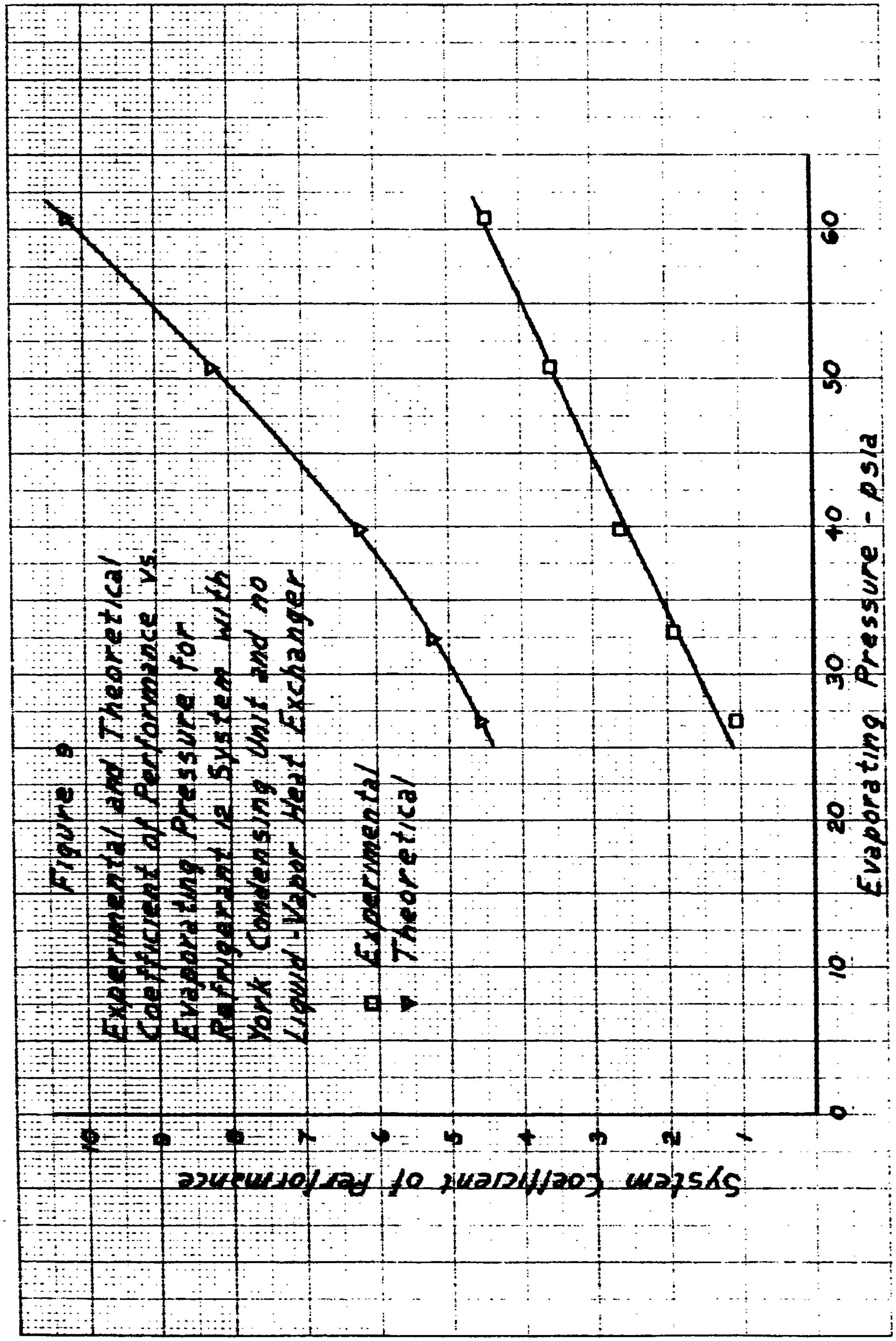

Figure 9
Experimental and Theoretical Coefficient of Performance vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit and no Liquid-Vapor Heat Exchanger

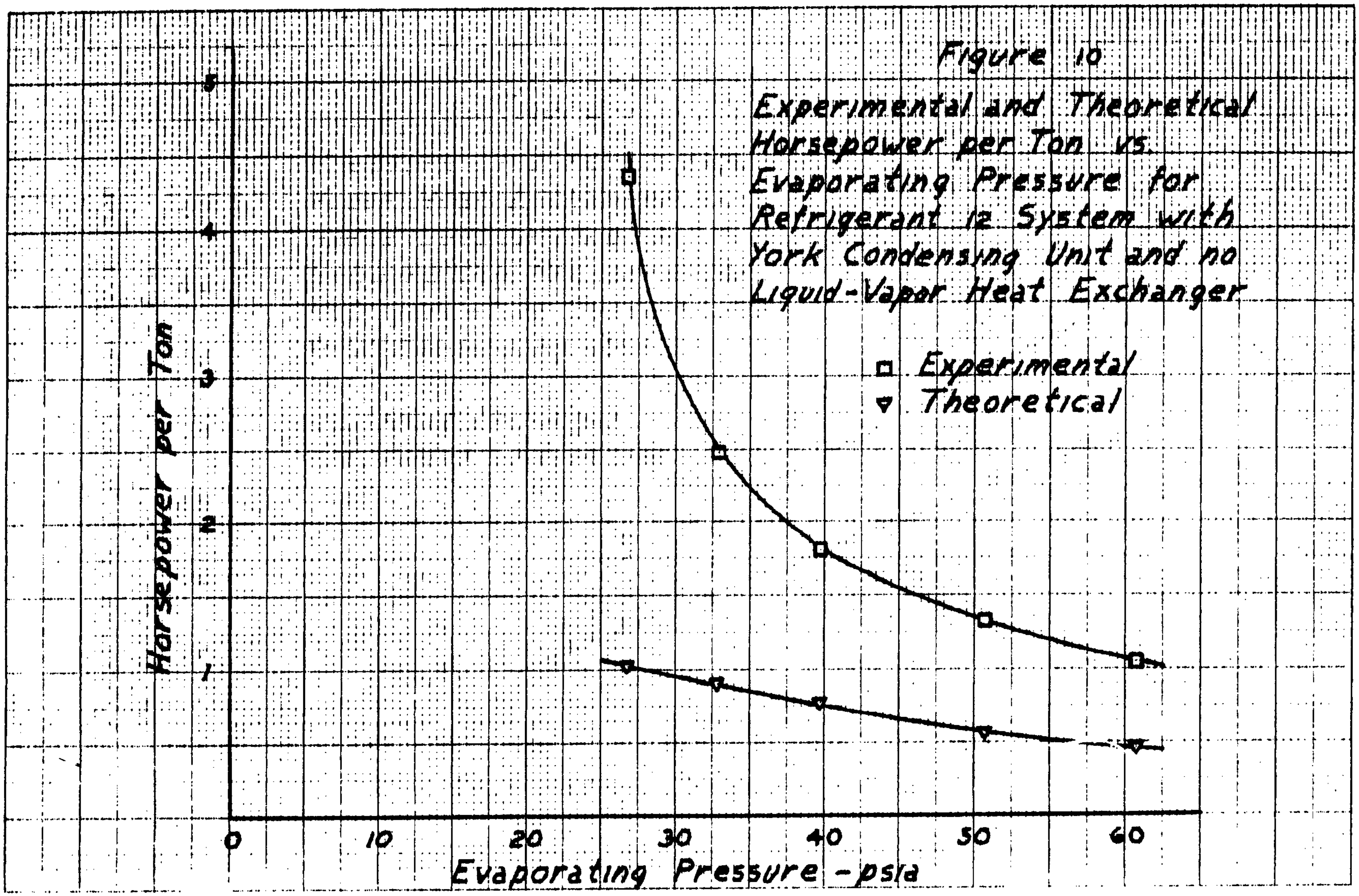

Figure 10

Experimental and Theoretical Horsepower per Ton vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit and no Liquid-Vapor Heat Exchanger

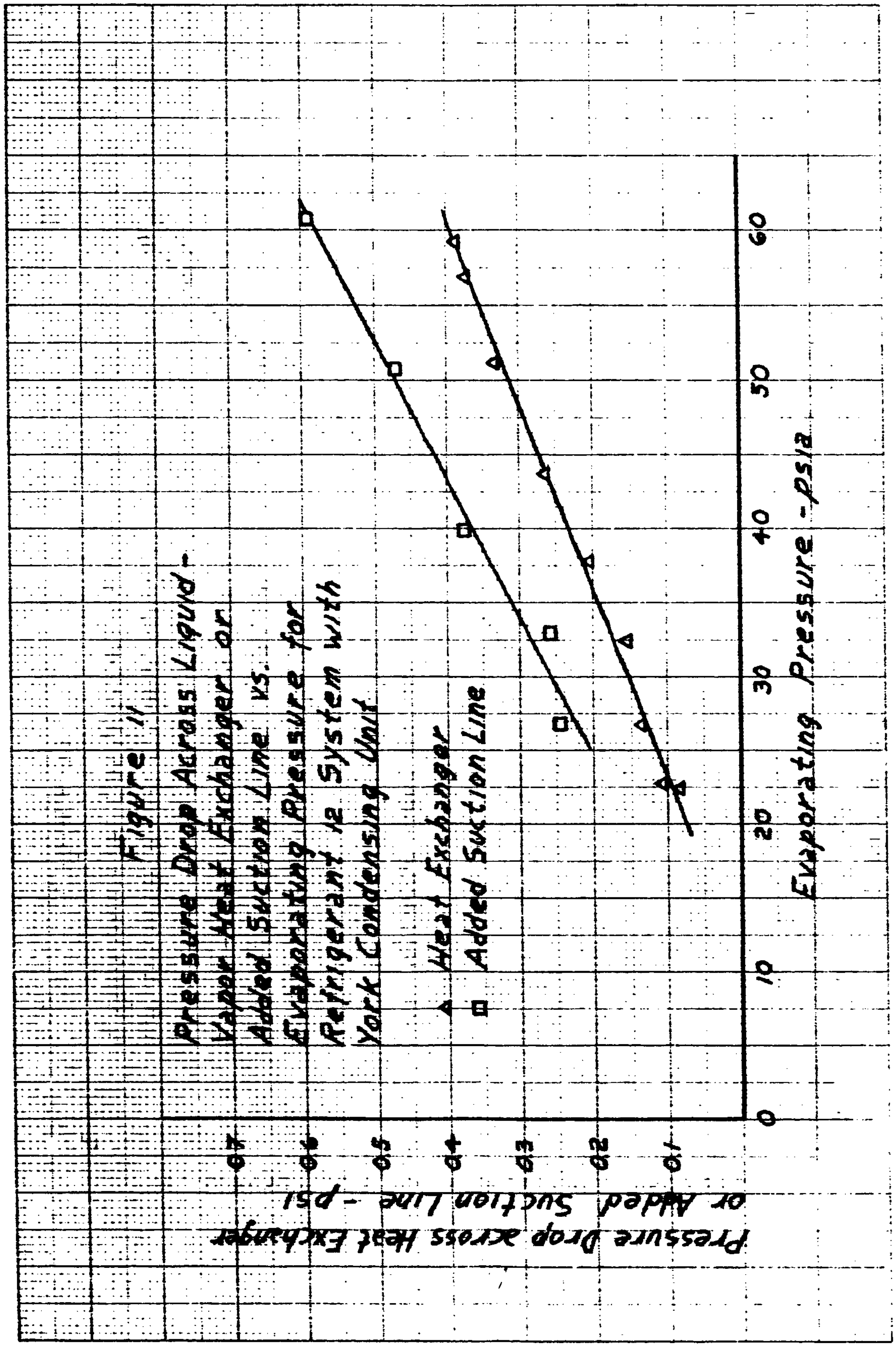

Figure 11
Pressure Drop Across Liquid-Vapor Heat Exchanger or Added Suction Line vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit

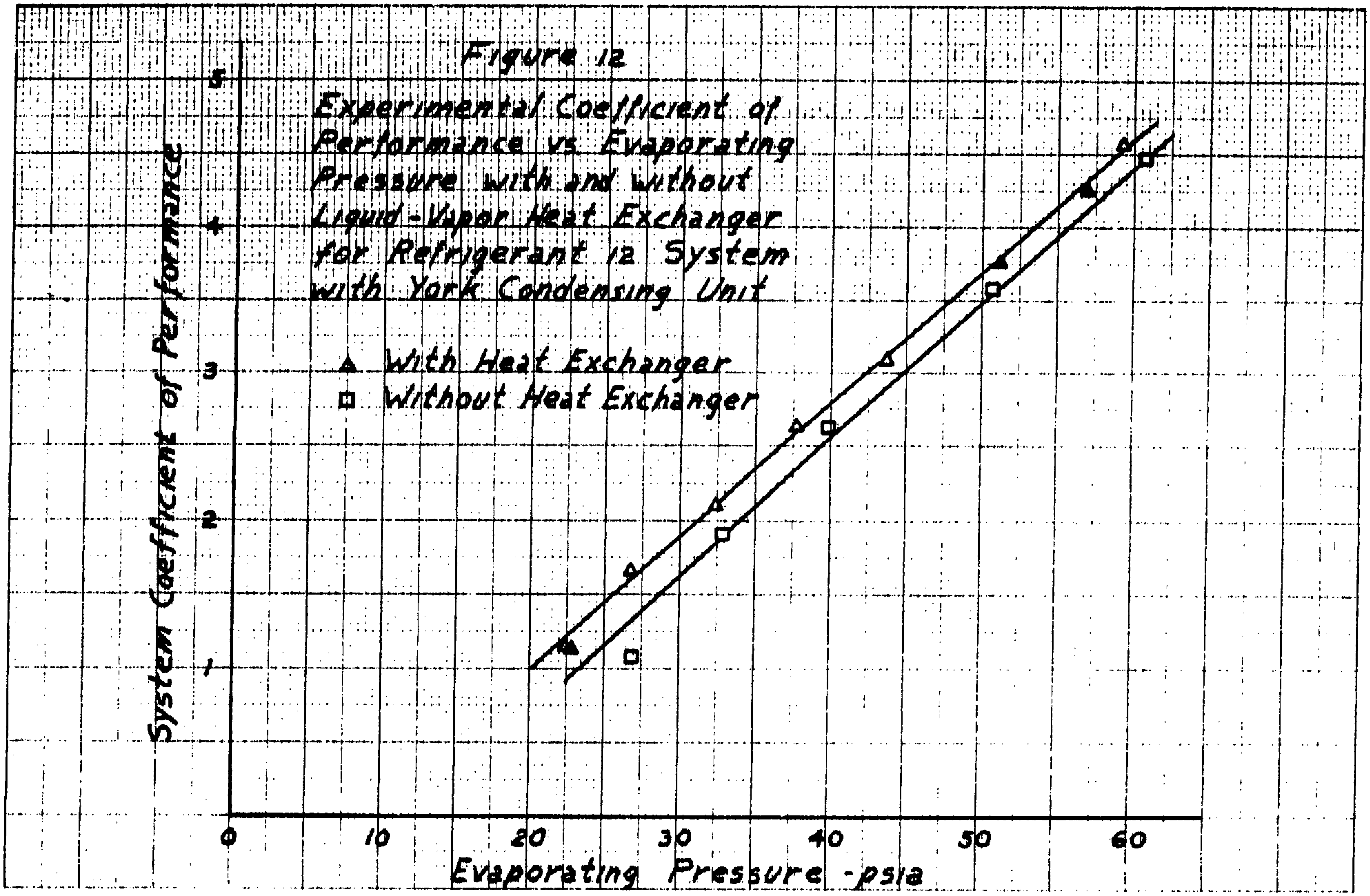

Figure 12

Experimental Coefficient of Performance vs Evaporating Pressure with and without Liquid-Vapor Heat Exchanger for Refrigerant 12 System with York Condensing Unit

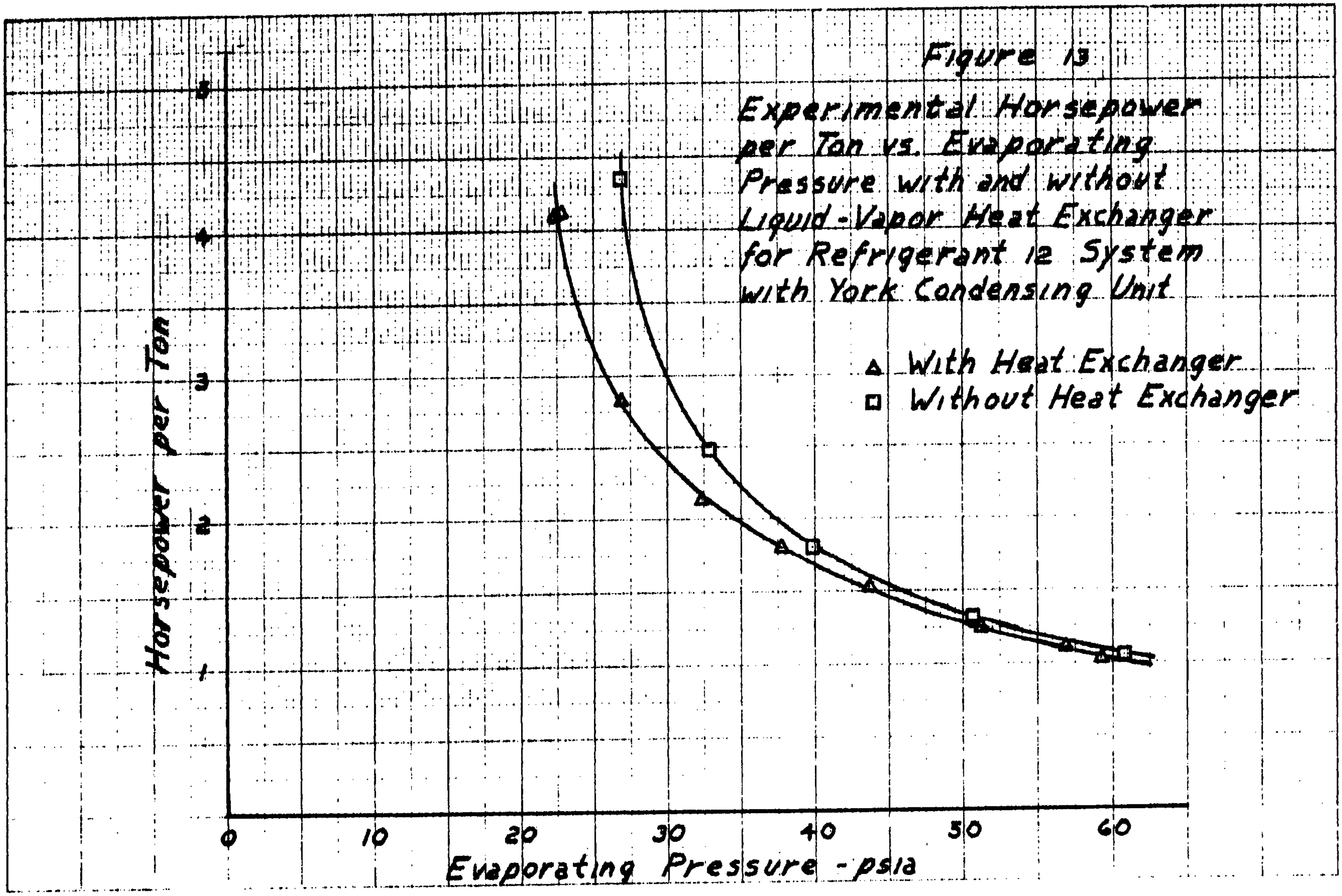

Figure 13

Experimental Horsepower per Ton vs. Evaporating Pressure with and without Liquid-Vapor Heat Exchanger for Refrigerant 12 System with York Condensing Unit

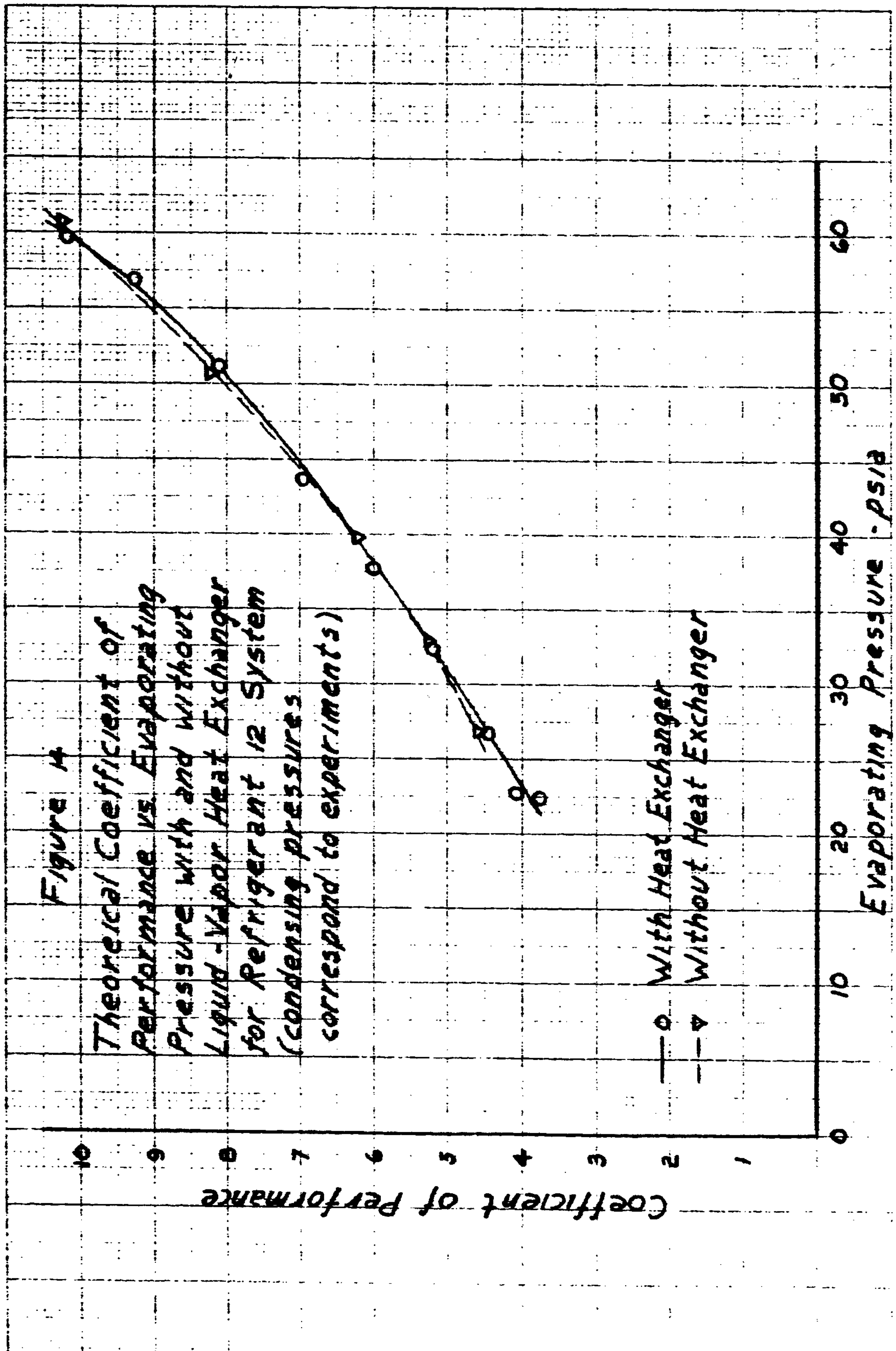

Figure 4

Theoretical Coefficient of Performance vs. Evaporating Pressure with and without Liquid-Vapor Heat Exchanger for Refrigerant 12 System (condensing pressures correspond to experiments)

V. CONCLUSIONS

From the calculations, and Figures 1, 2, and 14, it is obvious that installing a liquid-vapor heat exchanger for the purpose of increasing the COP is not worthwhile from a theoretical standpoint. Theoretically the use of a liquid-vapor heat exchanger will increase the capacity of a given evaporator at a given evaporating pressure.

For the laboratory system which was tested the experimental values of system COP were between 20% and 45% of the theoretical values. This range of values holds for both sets of runs, with and without the liquid-vapor heat exchanger. Similarly the experimental values of system $\frac{HP}{ton}$ for the system tested were between 200% and 450% of the theoretical values. The maximum experimental system COP that can be obtained with this laboratory Refrigerant 12 system is approximately 50% of the theoretical value. The actual system COP deviates from the theoretical COP more at higher evaporating pressures while the actual system $\frac{HP}{ton}$ deviates from the theoretical $\frac{HP}{ton}$ more at lower evaporating pressures.

The actual liquid-vapor heat exchanger effectiveness decreases as mean temperature difference of the two fluids decreases and the flow rate increases. Theoretically the heat exchanger effectiveness is a function of geometry only.

For an operating system containing Refrigerant 12 with a relatively short suction line, the increase in actual system COP achieved by using a liquid-vapor heat exchanger would not warrant its installation. The longer the suction line is in an operating system, the greater the heat transfer will be between the ambient air and the vapor, other conditions being constant. Therefore the advantage of a liquid-vapor heat exchanger to subcool the liquid refrigerant increases as the length of the suction line increases. The installation of a liquid-vapor heat exchanger to increase system COP

might be warranted in a Refrigerant 12 system with a long suction line.

If the evaporator capacity of a Refrigerant 12 system must be increased at a given evaporation pressure, the desired increase might be obtained with a liquid-vapor heat exchanger rather than a larger evaporator.

VI. RECOMMENDATIONS

Future work with liquid-vapor heat exchangers to subcool Refrigerant 12 should be limited to systems with long suction lines and to systems where it is desired to increase a given evaporator capacity. The increase in performance that can be obtained by installing a liquid-vapor heat exchanger in a system with a long suction line should be obtained.

The use of liquid-vapor heat exchangers and other methods of subcooling liquid refrigerant to increase system capacity could be compared. This comparison should be made on an economic basis as well as a mechanical performance basis.

VII. APPENDICES

APPENDIX A

SAMPLE THEORETICAL CALCULATION

Conditions and Assumptions

1. Vapor compression cycle with expansion valve
2. Refrigerant 12
3. No pressure drops in pipe lines, evaporator, or compressor manifolds and valves
4. Saturated vapor leaving evaporator
5. Saturated liquid leaving condenser
6. Isentropic compression
7. $p_E = 26$ psia
8. $p_c = 110$ psia

For a cycle with a liquid-vapor heat exchanger set the temperature of the liquid leaving the heat exchanger at 40° F.

$p_E = 26$ psia $\qquad h_{vi} = 77.710$ Btu/lb

$p_c = 110$ psia $\qquad h_{li} = h_{LCO} = 28.059$ Btu/lb

$t_{lo} = 40$ F $\qquad h_{lo} = 17.273$ Btu/lb

An energy balance around the heat exchanger fields:

$$h_{vo} = h_{vi} + (h_{li} - h_{lo}) = 77.710 + (28.059 - 17.273) = 88.496 \text{ Btu/lb}$$

$h_s = h_{vo} = 88.496$ Btu/lb

$p_{vo} = p_E = 26$ psia $\qquad s_{vo} = 0.19008$ Btu/lb°R

$s_d = s_s = s_{vo} = 0.19008$ Btu/lb°R $\qquad p_d = 110$ psia

$h_d = 101.551$ Btu/lb

$h_{LEX} = h_{lo} = 17.273$ Btu/lb

$$COP = \frac{h_{EE} - h_{LEX}}{h_d - h_s} = \frac{77.710 - 17.273}{101.551 - 88.496} = 4.6294$$

$$\frac{HP}{ton} = \frac{w(h_d - h_s)}{42.42} \; \frac{200}{w(h_{VE} - h_{LBX})} = \frac{4.714}{COP}$$

$$\frac{HP}{ton} = \frac{4.714}{4.6294} = 1.0182 \text{ hp/ton}$$

For a cycle without a heat exchanger $t_{lo} = t_{li} = t_{LCO}$ and for a cycle with an infinite counterflow heat exchanger $t_{vo} = t_{li}$.

The results used for Figures 1 and 2 are:

t_{lo}	COP	$\frac{HP}{ton}$	
87.23	4.5455	1.0370	(no heat exchanger)
80	4.5563	1.0346	
70	4.5693	1.0316	
60	4.5851	1.0281	
50	4.6072	1.0231	
40	4.6294	1.0182	
33.24	4.6467	1.0144	(Infinite heat exchanger)

APPENDIX B

EQUIPMENT DESCRIPTION AND SPECIFICATIONS

York Condensing Unit

Maker: York Corporation, York, Pennsylvania

Type: 422 FW

Speed: 375 rpm

Rating at 100 F Condensing Temperature:

a. 4890 Btu per hour at 20 F evaporator temperature

b. 7900 Btu per hour at 40 F evaporator temperature

Maximum condensing pressure	138 psig
Motor	3/4 hp
Bore	2 5/8 in.
Stroke	1 1/4 in.
Number of cylinders	2
Piston displacement per revolution	0.00782 cu. ft.
Refrigerant	Freon--12
Normal Refrigerant charge	7 lbs.
Receiver refrigerant capacity	40 lbs.
Crank case oil charge	5 pints

Control equipment:

Condenser water:	Penn Electric--Type XLI
Pressure safety:	Minneapolis-Honeywell-- Type L-413-1

Evaporator:

Type--Primary refrigerant calorimeter (flooded evaporator)

Insulation--Wood-covered cork, heat transfer factor 1.315 Btu/hr--°F

Expansion Valve:

Maker: Detroit Lubricator

Type: Thermostatic

Number: 783

Setting: ± 10°F superheat

Heat Exchanger:

Maker: Heat Exchanger Company, Incorporated, Brewster, New York

Model Number: 75-x

Wattmeters

Motor:	0-2000 watts
Evaporator heater:	0-8000 watts

Copper constantan thermocouple

Potentiometer--Leads and Northrop

Bourdon gages

Evaporator exit:	0-100 psig	
Vapor into heat exchanger:	0-200 psig	0-30 in. Hg vacuum
Vapor out of heat exchanger:	0-200 psig	0-30 in. Hg vacuum
Suction:	0-150 psig	0-30 in. Hg vacuum
Discharge:	0-150 psig	0-30 in. Hg vacuum

Mercury Manometer

APPENDIX C

TABLE I

TABULATED EXPERIMENTAL DATA

Date	4/28/59	4/30/59	4/30/59	5/2/59	5/2/59	5/6/59	5/6/59
Run Number	1	2	3	4	5	6	7
Duration of Test--min.	45	45	45	45	45	45	45
p_b--in. Hg	30.055	29.829	29.829	30.008	29.989	30.089	30.089
p_{EE}--psig	7.68	23.03	44.55	12.00	17.77	28.93	7.93
p_{vi}--psig	7.03	22.18	43.01	11.41	17.25	27.50	7.65
Δp_v--psi	0.085	0.24	0.39	0.14	0.16	0.26	0.11
p_s--psig	6.40	22.02	42.34	11.14	16.92	28.00	7.75
p_d--psig	97.04	100.75	102.20	100.90	101.65	101.50	96.20
t_{EE}--°F	0.6	30.0	58.8	8.4	21.8	37.5	2.6
t_{vi}--°F	4.9	31.4	59.4	12.6	24.1	36.8	7.5
t_{vo}--°F	47.6	44.3	64.1	44.1	41.6	47.4	36.5
t_s--°F	65.1	58.5	68.6	63.5	59.9	59.9	61.3
t_d--°F	157.4	136.4	121.9	154.2	144.6	131.1	154.7
t_{LCO}--°F	88.1	87.5	87.5	89.1	88.5	88.2	86.0
t_{li}--°F	87.1	84.7	85.4	87.7	86.0	85.9	84.8
t_{lo}--°F	84.5	70.7	80.0	73.5	68.8	76.7	83.2
t_{LBX}--°F	85.9	70.6	79.8	73.1	68.4	75.9	84.3
W_M--watts	630	726	736	674	714	734	635
W_H--watts	724	1905	3360	1120	1574	2263	726

TABLE I (CONTINUED)

Date	5/12/59	5/12/59	5/14/59	5/14/59	5/15/59		
Run Number	8	9	10	11	12	13	14
Duration of Test--min	45	45	45	45	45	45	45
P_b-- in. Hg	29.937	29.882	29.507	29.684	29.637	30.060	30.077
p_{HE}--psig	36.42	42.27	12.36	18.33	25.23	35.89	45.97
p_{vi}--psig	35.18	41.07	12.00	17.83	25.92	34.95	45.09
Δp_v--psi	0.33	0.37	0.25	0.26	0.38	0.47	0.59
p_s--psig	34.85	40.35	7.99	17.9	24.45	34.30	43.40
p_d--psig	104.55	105.35	97.35	100.6	102.8	102.40	105.40
t_{HE}--°F	48.1	56.0	8.2	16.9	29.0	45.9	60.1
t_{vi}--°F	65.6	56.8	-----	-----	-----	-----	-----
t_{vo}--°F	49.3	60.6	-----	-----	-----	-----	-----
t_s--°F	53.8	69.0	55.1	51.3	52.1	57.6	65.7
t_d--°F	128.9	126.6	145.8	136.1	129.4	120.9	149.5
t_{LCO}--°F	89.6	89.7	87.0	89.1	89.2	88.1	90.4
t_{li}--°F	88.3	88.3	-----	-----	-----	-----	-----
t_{lo}--°F	81.9	83.0	-----	-----	-----	-----	-----
W_M--watts	742	746	671	711	729	742	743
W_H--watts	2794	3172	726	1354	1913	2654	3322

APPENDIX D

SAMPLE EXPERIMENTAL CALCULATION

Run number 2 is used for an example.

Assumptions for calculations:

1. $p_{EE} = p_E$
2. $p_d = p_c$
3. Heat losses from the calorimeter are negligible.

$p_b = 29.829$ in. Hg $= 29.829\ (.4912)$ psia $= 14.67$ psia

$p_E = p_{EE} = 23.03$ psig $= 37.70$ psia

$t_E = 22.79$ F

$p_c = p_d = 100.75$ psig $= 115.42$ psia

$t_c = 90.64$ F

$$(COP)_A = \frac{W_H}{W_M} = \frac{1905}{726} = 2.63$$

$$\left(\frac{hp}{ton}\right)_A = \frac{4.714}{(COP)_A} = \frac{4.714}{2.63} = 1.80 \text{ hp/ton}$$

$Q_E = 3.413\ W_H = 3.413\ (1905) = 6500$ Btu/hr

$$\epsilon = \frac{t_{vo} - t_{vi}}{t_{li} - t_{vi}} = \frac{44.3 - 31.4}{70.7 - 31.4} = 0.242$$

$p_{vi} = 22.18$ psig $= 36.85$ psia $\qquad t_{vi} = 31.4$ F $\qquad h_{vi} = 81.003$ Btu/lb

$p_{vo} = p_{vi} - \Delta p_v = 36.85 - .21 = 26.64$ psia $\qquad t_{vo} = 44.3$ F $\qquad h_{vo} = 82.963$ Btu/lb

$t_{li} = 84.7 \qquad h_{li} = 27.464$ Btu/lb

$t_{lo} = 70.65 \qquad h_{lo} = 24.200$ Btu/lb

$(\Delta h_l - \Delta h_v)_{he} = (h_{li} - h_{lo}) - (h_{vo} - h_{vi}) = 1.334$ Btu/lb

For $(COP)_T$ and $(\frac{hp}{ton})_T$

$p_{EE} = 37.70$ psia $t_{EE} = 22.79$ F $h_{EE} = 79.675$ Btu/lb

$t_{LCO} = t_c = 90.637$ F $h_{LCO} = 28.864$ Btu/lb

From the definition of $\epsilon = \frac{t_{vo} - t_{vi}}{t_{li} - t_{vi}}$

$$t_{vo} = t_{vi} + \epsilon (t_{li} - t_{vi})$$

$$t_{vo} = 22.79 + .242(90.64 - 22.79) = 39.19 \text{ F}$$

$p_{vo} = p_E = 37.70$ $h_{vo} = 82.178$ Btu/lb $s_{vo} = 0.17205$ Btu/lb°R

$s_d = s_s = s_{vo} = 0.17205$ Btu/lb°R $p_d = 115.42$ psia

$h_d = 117.70$ Btu/lb

From energy balance around heat exchanger

$$h_{lo} = h_{li} - (h_{vo} - h_{vi}) = 28.864 - (82.158 - 79.675) = 26.381 \text{ Btu/lb}$$

$$h_{LBX} = h_{lo} = 26.381 \text{ Btu/lb}$$

$$(COP)_T = \frac{h_{EE} - h_{LBX}}{h_d - h_s} = \frac{79.675 - 26.381}{91.040 - 82.158} = 6.00$$

$$(\frac{hp}{ton})_T = \frac{4.714}{COP} = \frac{4.714}{6.00} = 0.787 \text{ hp/ton}$$

To find equivalent length of pipe for p across the heat exchanger

$h_{EE} = 80.768$ Btu/lb

$h_{LBX} = 24.189$ Btu/lb

$$w = \frac{3.413 W_H}{(h_{EE} - h_{LBX})60} = \frac{3.413 (1905)}{56.579 (60)} = 1.92 \text{ lb/min}$$

$v_{avg} = v$ at 36.7 psia and 37.9 F

From equation for $p = \frac{0.0121 fLw^2v}{d^5}$

$$L = \frac{pvd^5}{0.0121fw^2v_{avg}}$$

From Figure 6.6, Severns and Fellows (4) f= 0.02

From Figure 10.2, Severns and Fellows (4) d = 0.545 in. for 5/8 in. refrigeration tubing.

Solving for L yields L = 11.5 ft.

The calculations for runs without a heat exchanger are the same as this sample calculation except that $\epsilon = 0$. For runs without the heat exchanger no equivalent pipe length L was calculated.

APPENDIX E
TABLE II
TABULATED RESULTS

Run Number	1	2	3	4	5	6	7
p_E--psia	22.45	37.70	59.22	26.74	32.49	43.71	22.71
t_E--^oF	-2.84	22.79	47.87	5.47	15.12	30.70	-2.30
p_c--psia	111.81	115.42	116.87	115.64	116.37	116.28	110.98
t_c--^oF	88.36	90.64	91.43	90.69	91.13	91.08	87.84
$(COP)_A$	1.15	2.63	4.57	1.[illegible]6	2.21	3.09	1.14
$(\frac{hp}{ton})_A$--hp/ton	4.10	1.80	1.03	2.84	2.14	1.53	4.12
$(COP)_T$	3.76	6.00	10.19	4.45	5.20	6.[illegible]	4.09
$(\frac{hp}{ton})_T$--hp/ton	1.250	0.789	.463	1.060	0.917	0.677	1.151
Q_E--Btu/hr	2450	6500	11470	3820	5370	7740	2480
	0.519	0.242	0.179	0.380	0.283	0.182	0.376
h_{vi}--Btu/lb	84.298	81.033	84.157	7[illegible].935	80.23[illegible]	81.82[illegible]	78.451
h_{vo}--Btu/lb	28.017	82.963	84.922	85.1[illegible]	82.959	83.156	82.651
h_{li}--Btu/lb	27.406	27.464	27.628	2[illegible].1[illegible]8	27.757	27.746	27.476
h_{lo}--Btu/lb	27.406	24.200	26.353	24.[illegible]7	23.764	25.597	27.100
$(\Delta h_l - \Delta h_v)_{hc}$--Btu/lb	-5.565	1.334	0.510	-2.382	0.473	0.819	-3.824

TABLE II (CONTINUED)

Run Number	8	9	10	11	12	13	14
p_E--psia	51.12	56.95	26.86	32.92	39.81	50.67	60.75
t_E--^oF	39.36	45.5[illegible]	5.68	15.77	25.67	38.90	49.38
p_c--psia	119.25	120.03	111.85	115.19	117.38	117.18	120.18
t_c--^oF	92.85	93.34	88.38	90.42	91.74	91.62	93.39
$(COP)_A$	3.76	4.25	1.0[illegible]	1.91	2.63	3.5[illegible]	4.48
$(\frac{hp}{ton})_A$--hp/ton	1.25	1.11	4.35	2.48	1.30	1.32	1.05
$(COP)_T$	8.10	9.27	4.59	5.22	5.21	[illegible].22	10.22
$(\frac{hp}{ton})_T$--hp/ton	0.[illegible]83	0.509	1.[illegible]23	0.905	0.770	[illegible].574	0.462
Q_E--Btu/hr	9520	10810	2480	4620	6530	9050	11330
	0.115	0.121	----	----	----	----	----
h_{vi}--Btu/lb	83.006	83.853	----	----	----	----	----
h_{vo}--Btu/lb	83.730	84.479	----	----	----	----	----
h_{li}--Btu/lb	2[illegible].312	28.312	----	----	----	----	----
h_{lo}--Btu/lb	26.809	27.065	----	----	----	----	----
$(\Delta h_l - \Delta h_v)_{he}$--Btu/lb	0.779	0.621	----	----	----	----	----

APPENDIX I

NOMENCLATURE

d	diameter
f	friction factor
h	enthalpy
L	equivalent length of pipe
p	pressure
Q_E	evaporator capacity
s	entropy
t	temperature in degrees Fahrenheit
v	specific volume
w	flow rate
W	Watts
COP	coefficient of performance
$\frac{hp}{ton}$	horsepower per ton
Δ	finite difference
ϵ	heat exchanger effectiveness

SUBSCRIPTS

A	actual or experimental
b	barometer
c	conditions in condenser
d	discharge from compressor
E	conditions in evaporator
EE	evaporator exit

H	calorimeter heater
he	heat exchanger
LBX	liquid refrigerant before expansion valve
LCO	liquid out of condenser
li	liquid into heat exchanger
lo	liquid out of heat exchanger
M	compressor motor
s	suction at compressor
T	theoretical
vi	vapor into heat exchanger
vo	vapor out of heat exchanger

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